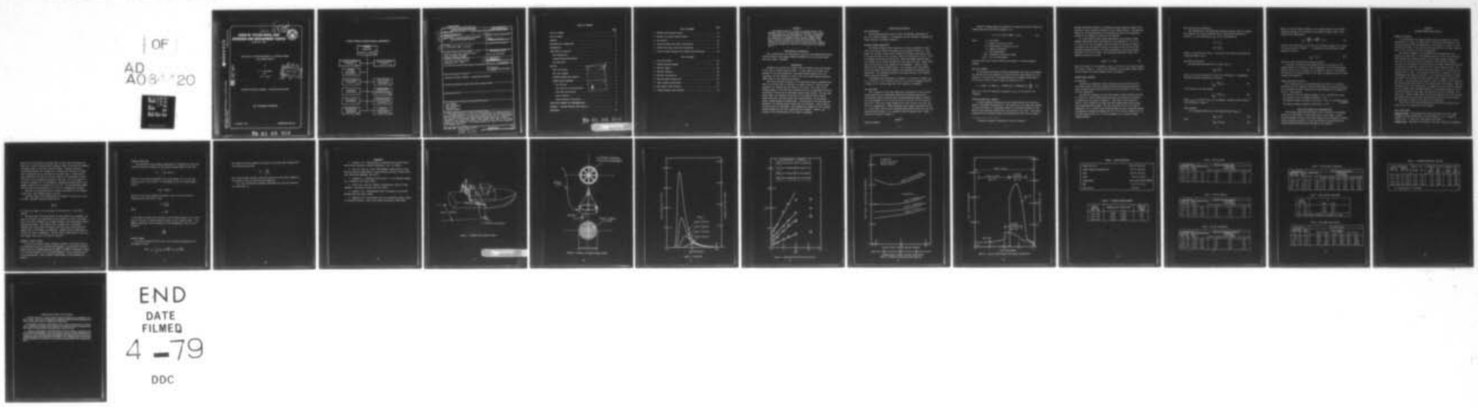
AD-A064 420 DAVID W TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CE--ETC F/G 13/10
PREDICTION OF THE EFFECTIVENESS OF FLOPPER-STOPPER ROLL DAMPING--ETC(U)
NOV 78 W R MCCREIGHT, H D JONES

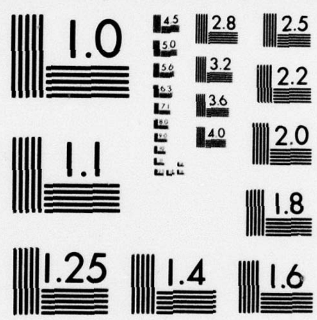
UNCLASSIFIED DTNSRDC/SPD-0835-01 NL

| OF |

AD
A064420



END
DATE
FILMED
4-79
DDC


1.0
4.5
5.0
5.6
6.3
2.8
3.2
3.6
4.0
2.5
2.2
2.0
1.1
1.8
1.25
1.4
1.6

LEVEL II (12) Nu

AD A064420

DTNSRDC/SPD-0835-01

DAVID W. TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

Bethesda, Md. 20084

PREDICTION OF THE EFFECTIVENESS OF A FLOPPER-STOPPER ROLL DAMPING DEVICE

by

W. R. McCreight

and

H. D. Jones

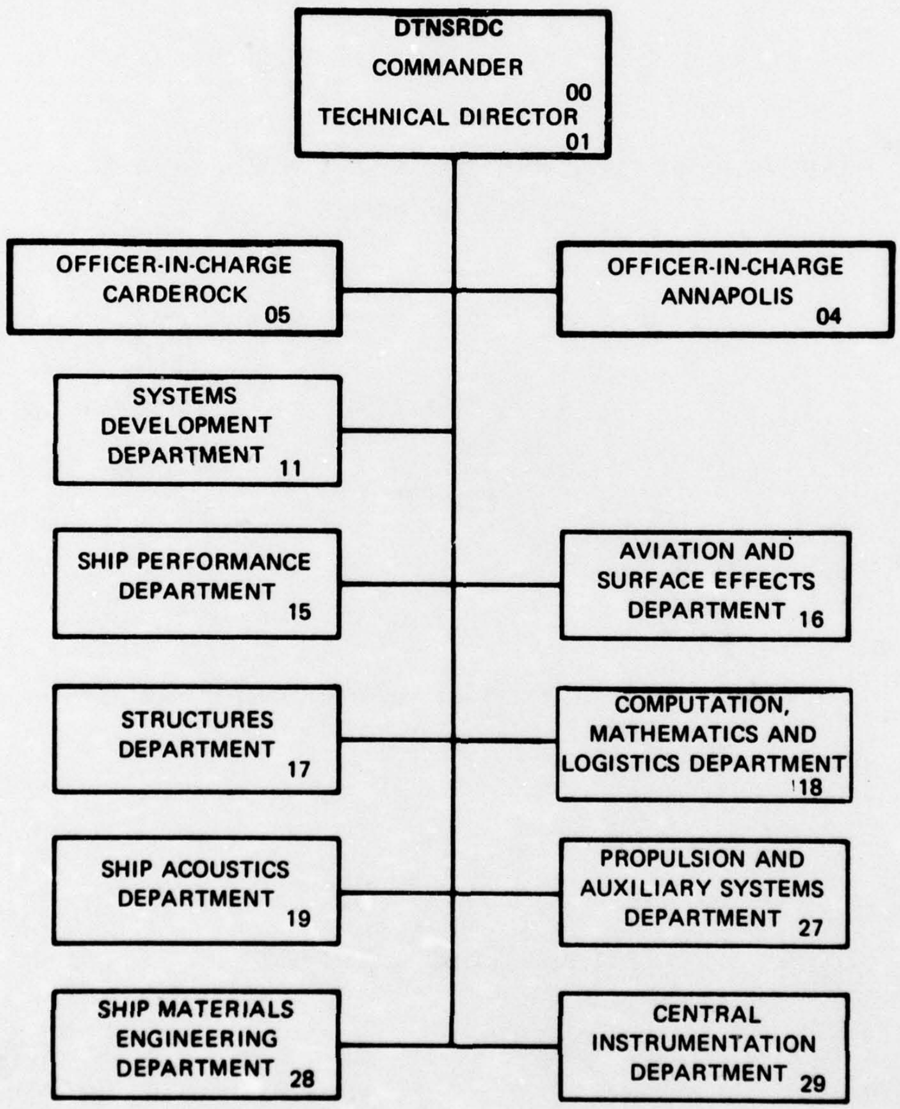
APPROVED FOR PUBLIC RELEASE: DISTRIBUTION UNLIMITED

SHIP PERFORMANCE DEPARTMENT

November 1978 DTNSRDC/SPD-0835-01

PREDICTION OF THE EFFECTIVENESS OF A FLOPPER-STOPPER ROLL DAMPING DEVICE

MAJOR DTNSRDC ORGANIZATIONAL COMPONENTS



UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE

READ INSTRUCTIONS BEFORE COMPLETING FORM

1. REPORT NUMBER
14 DTNSRDC/SPD-0835-01

2. GOVT ACCESSION NO.

3. RECIPIENT'S CATALOG NUMBER

4. TITLE (and Subtitle)
6 PREDICTION OF THE EFFECTIVENESS OF FLOPPER-STOPPER ROLL DAMPING DEVICE.

5. TYPE OF REPORT & PERIOD COVERED
9 Final rept.

6. PERFORMING ORG. REPORT NUMBER

7. AUTHOR(s)
10 W. R. McCreight [redacted] H. D. Jones

8. CONTRACT OR GRANT NUMBER(s)

9. PERFORMING ORGANIZATION NAME AND ADDRESS
David W. Taylor Naval Ship R&D Center
Ship Performance Department
Bethesda, Maryland 20084

10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
Work Unit 1-1568-890

11. CONTROLLING OFFICE NAME AND ADDRESS
Naval Ship Engineering Center
Washington, D.C. 20362

12. REPORT DATE
11 Nov[redacted]1978

13. NUMBER OF PAGES
26 12 28p.

14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)

15. SECURITY CLASS. (of this report)
Unclassified

15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

APPROVED FOR PUBLIC RELEASE: DISTRIBUTION UNLIMITED

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Roll Damping
Roll Predictions
Ship Motions
"Flopper-Stopper"

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The effectiveness of a series of flopper-stopper roll damping devices on a Torpedo Weapons Retriever at zero speed in random beam seas was investigated. The ship was modeled as a one-degree-of-freedom system, and the flopper-stopper was modeled as an equivalent linear damping element based upon a detailed analysis of the forces on the device. The tension in the cable of the flopper-stopper was also estimated. The flopper-stopper was found to be an effective roll damping device.

DD FORM 1 JAN 73 1473 EDITION OF 1 NOV 65 IS OBSOLETE
S/N 0102-LF-014-6601

389 694

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

JOB

TABLE OF CONTENTS

Page

LIST OF FIGURES

LIST OF TABLES

ABSTRACT

The effectiveness of a series of flopper-stopper roll damping devices on a Torpedo Weapons Retriever at zero speed in random beam seas was investigated. The ship was modeled as a one-degree-of-freedom system, and the flopper-stopper was modeled as an equivalent linear damping element based upon a detailed analysis of the forces on the device. The tension in the cable of the flopper-stopper was also estimated. The flopper-stopper was found to be an effective roll damping device.

ADMINISTRATIVE INFORMATION

This investigation was authorized by the Naval Ship Engineering Center, Norfolk Division, Work Request N64281-78-WR80045 and was performed under Work Unit Number 1-1568-890.

INTRODUCTION

Excessive roll motion of a ship at sea can severely limit the ability of the crew to perform required tasks. In this report the results of an analysis of the effectiveness of a proposed flopper-stopper device for increasing the roll damping of a 100 ft (30.5 m) Torpedo Weapons Retriever (TWR) vessel at zero speed are presented.

The flopper-stopper consists of two drogue-like devices suspended from booms, one on each side of the ship, as shown in Figure 1, which provide increased roll damping and thus reduce roll motion. The forces on the flopper-stopper as the ship moves through one cycle of sinusoidal roll were calculated in detail and an equivalent linear model derived. The root mean square (RMS) roll response of the ship at zero speed in short-crested random beam seas, with and without the flopper-stopper, was calculated using a single-degree-of-freedom model. Acceleration at the deck edge and tensions in the flopper-stopper cable were also analyzed. Shock loads in the cable were not considered. The analysis was repeated for four flopper-stopper designs and for four significant wave heights. Finally, the validity of the idealizations in the model is examined.

CONDITIONS FOR ANALYSIS

SHIP DESCRIPTION

The ship analyzed was the 100 ft (30.5 m) TWR-681, particulars of which are listed in Table 1. An isometric view of the ship, including the flopper-stopper, is shown in Figure 1.

FLOPPER-STOPPER DESCRIPTION

The flopper-stopper device consists of a bucket suspended by cable from a boom on each side of the ship, as may be seen in Figure 1. Each bucket is a hollow, vertical axis cylinder of 10 pound steel plate, open at the top and with a 4-in. by 4-in. (10 cm x 10 cm) grid of 1/2 in. (1.25 cm) diameter steel bars on the bottom. Eight pie-shaped neoprene flaps are attached to the inside cylinder wall just above the grid. These check valves allow water to flow upwards but not downwards. Figure 2 shows details of the flopper-stopper bucket. As the ship rolls, the buckets move up and down, the valves closing on the upstroke and opening on the downstroke. The resulting tensions in the cable produce a roll damping moment acting on the ship.

Calculations were performed for buckets 1 ft (0.30 m) in height, and 2.5 ft (0.76 m), 3.2 ft (0.98 m) and 4.5 ft (1.37 m) in diameter; and for a bucket 3.0 ft (0.91 m) high, and 3.2 ft (0.98 m) in diameter.

SEA CONDITIONS

Roll responses at zero ship speed were calculated for short-crested random waves with the predominant waves from 90 degrees (beam seas). Bretschneider spectra was used with cosine-squared spreading and significant wave heights of 4.6 ft (1.40 m), 6.9 ft (2.10 m), 10.0 ft (3.05 m), and 15.0 ft (4.57 m) at modal periods of 6.2 sec, 8.5 sec, 10.5 sec, and 11.9 sec, respectively. These significant wave heights represent State 3, 4, 5, and 6 seas, with the most probable modal periods which occur for the northern North Atlantic, i.e., between 40 and 60 degrees north. These spectra are plotted in Figure 3.

ANALYSIS

SHIP ROLL MOTION

Conolly's* single-degree-of-freedom roll equation was used to model the unstabilized ship roll motion response, i.e.,

$$(I + \delta I)\ \ddot{\phi} + 2N\dot{\phi} + \Delta\overline{GM}_t\,\phi = F_\phi(t) \qquad (1)$$

where ϕ = roll angle

I = moment of inertia in roll

δI = added moment of inertia in roll

N = damping coefficient

Δ = ship displacement

$\overline{GM}_t$ = transverse metacentric height

$F_\phi(t)$ = roll exciting moment

Cox and Lloyd[2] have further developed this method to include nonlinear damping.

HULL ROLL DAMPING

By interpolating between available experimental bare hull roll damping for similar hulls and estimating the bilge keel and skeg damping from the formula of Cox and Lloyd,[2] the nondimensional roll damping coefficient of the TWR hull without flopper-stopper was estimated to be

$$n = 0.00657 + 0.00564\ \phi_a - 0.000191\ \phi_a^2 + 0.00000289\ \phi_a^3 = \frac{\omega_\phi N}{\Delta GM} \qquad (2)$$

where ϕ_a is the roll amplitude in degrees, and ω_ϕ is the natural roll frequency.

FLOPPER-STOPPER ROLL DAMPING

A linear damping coefficient for the flopper-stopper was obtained by means of equivalent linearization.[3] The upper end of the cable was assumed to move vertically in a sinusoidal motion at the natural frequency of the ship and the resultant tensions were calculated for one cycle. The energy dissipated by the flopper-stopper was calculated and the value of linear

*A complete listing of references is given on page 11.

damping coefficient required to dissipate the same energy was taken as the equivalent linear damping coefficient of the flopper-stopper. The inertial effects of the flopper-stopper were examined and found to be negligible. Details of the analysis are given in the appendix.

For each flopper-stopper design considered, the equivalent linear damping coefficients were calculated for roll amplitudes of 1, 2.5, 5, 7.5, 10, 16, 24, and 32 degrees, and the results are plotted in Figure 4. A straight line was fitted to each of the curves at the low end of the amplitude range and added to the above expression for the hull damping and input to the roll program. The calculated equivalent linear roll damping is given in the form

$$n_{FS}(\phi) = a + b_e \phi_a \tag{3}$$

where the values for the constants a and b_e for each of the flopper-stoppers are listed in Table 2. Table 2 also indicates the approximate upper limits of the validity for these damping curves, see Figure 4.

RANDOM SEAWAY RESPONSE

Roll Motions

The RMS roll responses in short-crested seas were calculated from wave height spectra and roll transfer functions derived from equation 1 using frequency domain methods described in Cox and Lloyd[2] for the specified sea conditions.

The RMS roll responses for the ship without flopper-stoppers and with the three 1-ft (0.30 m) high flopper-stoppers are plotted as a function of significant wave height in Figure 5. This data is also listed in Table 3, and it should be recalled that the assumed modal period varies with significant wave height. The roll motion for the 3.2 ft (0.98 m) diameter by 3.0 ft (0.91 m) high flopper-stopper was not calculated because the damping is between the damping for the 3.2 ft (0.98 m) diamter by 1 ft (0.30 m) high and the 4.5 ft (1.37 m) diameter by 1 ft (0.30 m) high flopper-stopper.

Roll Velocity and Acceleration

The assumption of a narrow-banded response spectrum, which is reasonable because roll motion is sharply tuned, implies that the RMS roll velocity, $\sigma_{\dot{\phi}}$, and acceleration, $\sigma_{\ddot{\phi}}$, can be approximated by

$$\sigma_{\dot{\phi}} = \omega_{\phi}\, \sigma_{\phi}$$

$$\sigma_{\ddot{\phi}} = \omega_{\phi}^{2}\, \sigma_{\phi}$$

where σ_{ϕ} is the RMS roll motion. The results from these calculations are listed in Tables 4 and 5.

Deck Edge Acceleration

The deck edge acceleration due to ship roll is

$$a_{DE} = \frac{B}{2}\, \ddot{\phi} \tag{4}$$

where B is the ship beam and $\ddot{\phi}$ is the roll acceleration. Consequently, the RMS deck edge acceleration is given by

$$\sigma_{a_{DE}} = \frac{B}{2}\, \omega_{\phi}^{2}\, \sigma_{\phi} \tag{5}$$

In the present case this becomes

$$\sigma_{a_{DE}} = 0.44\, \sigma_{\phi} \tag{6}$$

where $\sigma_{a_{DE}}$ is in ft/sec^2 and σ_{ϕ} is in degrees. Results of this calculation are given in Table 6.

Cable Tensions

If the damping moment due to the flopper-stopper, M_{FS}, is

$$M_{FS} = 2N\, \dot{\phi} \tag{7}$$

then

$$M_{FS} = 2R\, T_{FS} \tag{8}$$

where R is the horizontal distance of the flopper-stopper from the ship centerline and T_{FS} is the tension in the flopper-stopper cables. This tension can be expressed as

$$T_{FS} = \frac{n\Delta\overline{GM}}{\omega_\phi R}\dot{\phi} = \frac{n\Delta\overline{GM}}{R}\phi = C_T\,\phi \tag{9}$$

Values of the coefficient C_T for each of the flopper-stoppers at an angle of 5 deg are listed in Table 7. It follows that the RMS dynamic cable tension is given by

$$\sigma_{T_{FS}} = C_T\,\sigma_\phi \tag{10}$$

Results of this calculation are given in Table 8.

It should be noted that this analysis gives a low value for the tension because (1) the static load is neglected, (2) the actual dynamic tension curve is not a sinusoid, but is peaked as shown in the appendix, (3) shocks due to sudden closing of the flaps are neglected, and (4) cable dynamics are neglected.

Single Amplitude Statistics

For a process with a narrow-banded spectrum, which is a realistic assumption for roll response, various single amplitude response statistics can be obtained by simply multiplying the RMS response by an appropriate factor.[4] Specifically,

The average of the 1/3 highest responses = 2.0 σ (significant value)

The average of the 1/10 highest responses = 2.55 σ

The expected highest response in 50 cycles = 3.03 σ (a reasonable "peak" response)

The expected highest response in 500 cycles = 3.72 σ

CONCLUDING REMARKS AND RECOMMENDATIONS

The flopper-stopper has been shown to be an effective roll damping device for a TWR vessel or other small craft at zero speed. A more accurate estimate of motions and cable tensions can be obtained by numerically integrating equation 1 in the time domain and incorporating the detailed flopper-stopper model described in the appendix.

APPENDIX

FLOPPER-STOPPER FORCE ANALYSIS

METHOD OF ANALYSIS

In this analysis the flopper-stopper system is modeled as a device suspended from a cable, the upper end of which moves sinusoidally in the vertical direction with an amplitude $R\phi_a$ and frequency ω, where R is the horizontal distance of the cable from the centerline of the ship, taken as 40.0 ft (12.2 m) and ϕ_a is the roll amplitude.

The cable is assumed to be inextensible and consequently cable dynamics are neglected. If the downward velocity of the cable suspension point exceeds the terminal velocity of the flopper-stopper bucket in a free fall in water, the bucket is assumed to fall with this terminal velocity with no tension in the cable until the bucket reaches the position it would have reached had it moved with the cable suspension point. As the bucket moves up, the flappers will be closed and as it moves down the flappers will be open. The transition either way is assumed to occur instantaneously and without shock load. The transition from open to closed is assumed to occur when the bucket begins to move upward at the end of the cycle. The transition from closed to open is assumed to occur when the net drag and added mass forces on the closed bucket are positive. That is, the drag force on the closed bucket is assumed to act as a uniform pressure on the (closed) flaps, the added mass force is likewise assumed to act as a uniform pressure on the flaps, and when the net force is upwards, the flaps open. Figure 6 shows plots of force and position for a case in which the line does not go slack and a second case in which the does go slack. The opening and closing of the flaps is also indicated.

The effects of the free surface and any mean current induced by the bucket motion are neglected in the analysis.

DRAG COEFFICIENTS

<u>Flappers Closed</u>. From Hoerner[5] the drag coefficient for a cup, rigid parachute or other similar body is in the range 1.0 to 1.4, based on frontal area. A value of 1.1 was used in this case.

<u>Flappers Open</u>. The drag on the bucket in the open condition is assumed to

be the sum of the drag for the bucket wall (a short open tube) plus the drag on a circular cylinder with a parameter equal to the grid bar diameter and length equal to the total grid bar length. Possible drag due to interference between the crossed bars or between the bars and the bucket wall is neglected, as is the effect of the open flaps. Hoerner gives a drag coefficient of 0.025 for the wall, based on an area equal to twice the product of the diameter and depth of the tube. At a roll angle of 5 deg and a frequency of 1.57 radians/sec, the Reynolds number based on the maximum velocity of the cylinder is 2.12×10^4. For this Reynolds number, Hoerner gives a drag coefficient of 1.2 for a two-dimensional cylinder.

The mass of the enclosed water is easily calculated for the flappers closed case. With flappers open it is zero.

The added mass of the closed device is assumed to be that of a circular disk. Kennard[6] quotes an added mass of

$$\frac{8}{3}\rho r_d^3$$

for this case, where r_d is the radius of the disk and ρ is the fluid density.

The added mass of the open device is quite small and is assumed to be equal to the sum of the added mass of the cylinder wall and of the added mass of the bars considered as a single cylinder, with interference effects neglected as before. The cylinder wall is modeled as a rectangular cylinder with cross section the same as the wall cross section and with length equal to the bucket circumference. Kennard gives an added mass of $0.0142\ \rho$ per unit circumference. The bars have an added mass of $\rho\pi r_b^2$ per unit length, where r_b is the radius of the bar.

SUMMARY OF DRAG AND MASS

For the four flopper-stopper designs considered, fluid drag force F_V and fluid mass force F_A, including enclosed mass, are presented in Table 9. These forces are nondimensionalized by $\rho\frac{\pi}{4}d^2V^2$ and $\rho\frac{\pi}{6}d^3A$ respectively, where ρ is the fluid density, d is the device diameter, V is the velocity and A is the acceleration. The air mass of each bucket is also presented in Table 9.

DAMPING COEFFICIENT

The equivalent linear damping coefficient is calculated by first computing the energy dissipated by the flopper-stopper moment in one cycle

$$E = \int_{o}^{T} M(t)\ \dot{\phi}(t)\ dt$$

where E is the energy dissipated, M(t) is the moment, $\dot{\phi}(t)$ is the roll velocity, and T is the period. If the moment is due to a linear damper, that is

$$M(t) = 2N\dot{\phi}(t)$$

where N is the linear damping coefficient, and if the roll motion is sinusoidal with amplitude A, then

$$N = \frac{E}{2\pi\omega A^2}$$

where

$$\omega = \frac{2\pi}{T}$$

It can easily be shown that the effect of the flopper-stopper, one on each side of the ship, is simply twice the effect of one flopper-stopper. The damping coefficient thus calculated is then nondimensionalized by multiplying by

$$\frac{\omega_{\phi}}{\Delta GM}$$

INERTIAL TERMS

By Fourier analysis of the moment, the following representation of the moment is found

$$M(t) = A_o + \sum_{n=1}^{\infty} [A_n \cos \frac{2\pi nt}{T} + B_n \sin \frac{2\pi nt}{T}]$$

The inertial force is assumed to be equal to the first term in phase with the acceleration and thus

$$M_a = -\frac{A_1}{\omega^2 \phi_a}$$

The flopper-stopper inertial terms were found to be very small compared to the inertia of the hull and thus were neglected.

Note that the equivalent damping coefficient can also be found from the coefficient B_1.

REFERENCES

1. Conolly, J.E., "Rolling and Its Stabilization by Active Fins," Trans. Royal Institute of Naval Architecture, Vol. 111 (1969).

2. Cox, G.G. and A.R. Lloyd, "Hydrodynamic Design Basis for Navy Ship Roll Motion Stabilization," Annual Meeting, Society of Naval Architects and Marine Engineers (Nov 1977).

3. Minorsky, N., "Nonlinear Oscillations," D. van Nostrand Company, Inc., Princeton, New Jersey (1962).

4. Price, W.G. and R.E.D. Bishop, "Probabilistic Theory of Ship Dynamics," John Wiley and Sons, New York (1974).

5. Hoerner, S.F., "Fluid-Dynamic Drag," Published by the Author, Midland Park, New Jersey (1965).

6. Kennard, E.H., "Irrotational Flow of Frictionless Fluids, Mostly of Invariable Density," David Taylor Model Basin Report 2299 (1967).

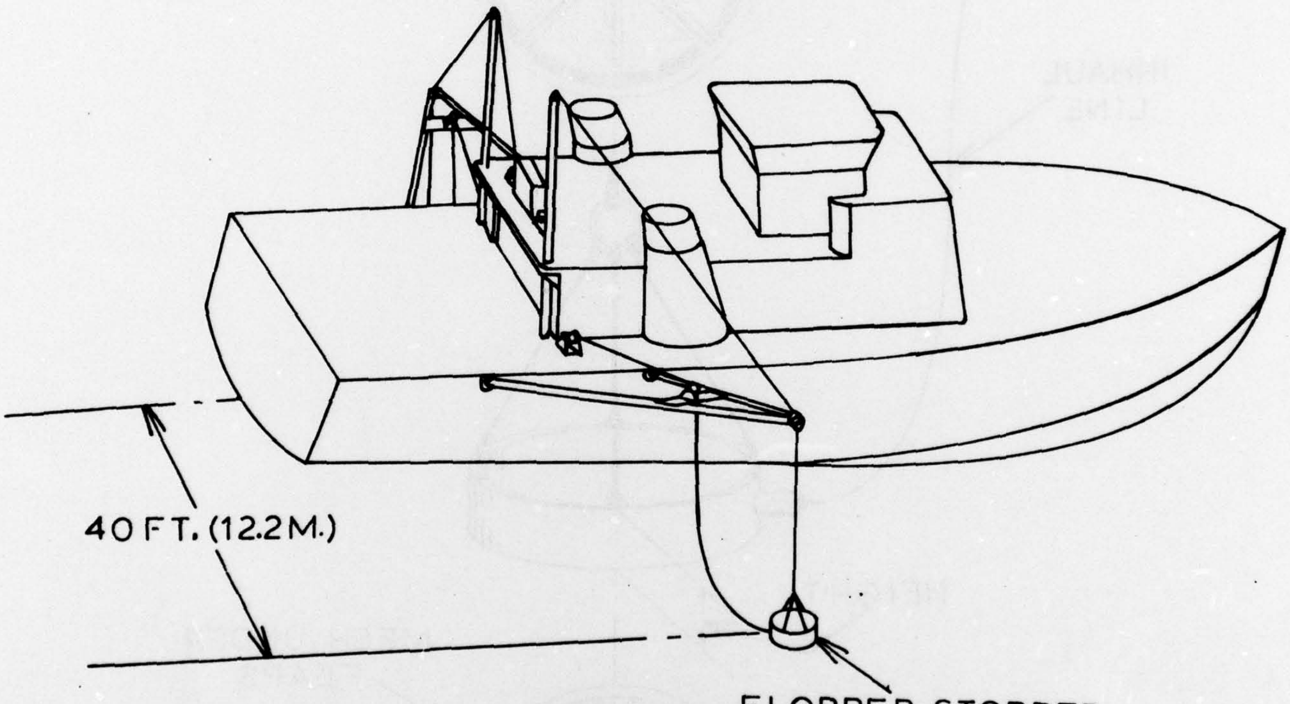


Figure 1 - TWR-681 with Flopper-Stopper

[redacted]

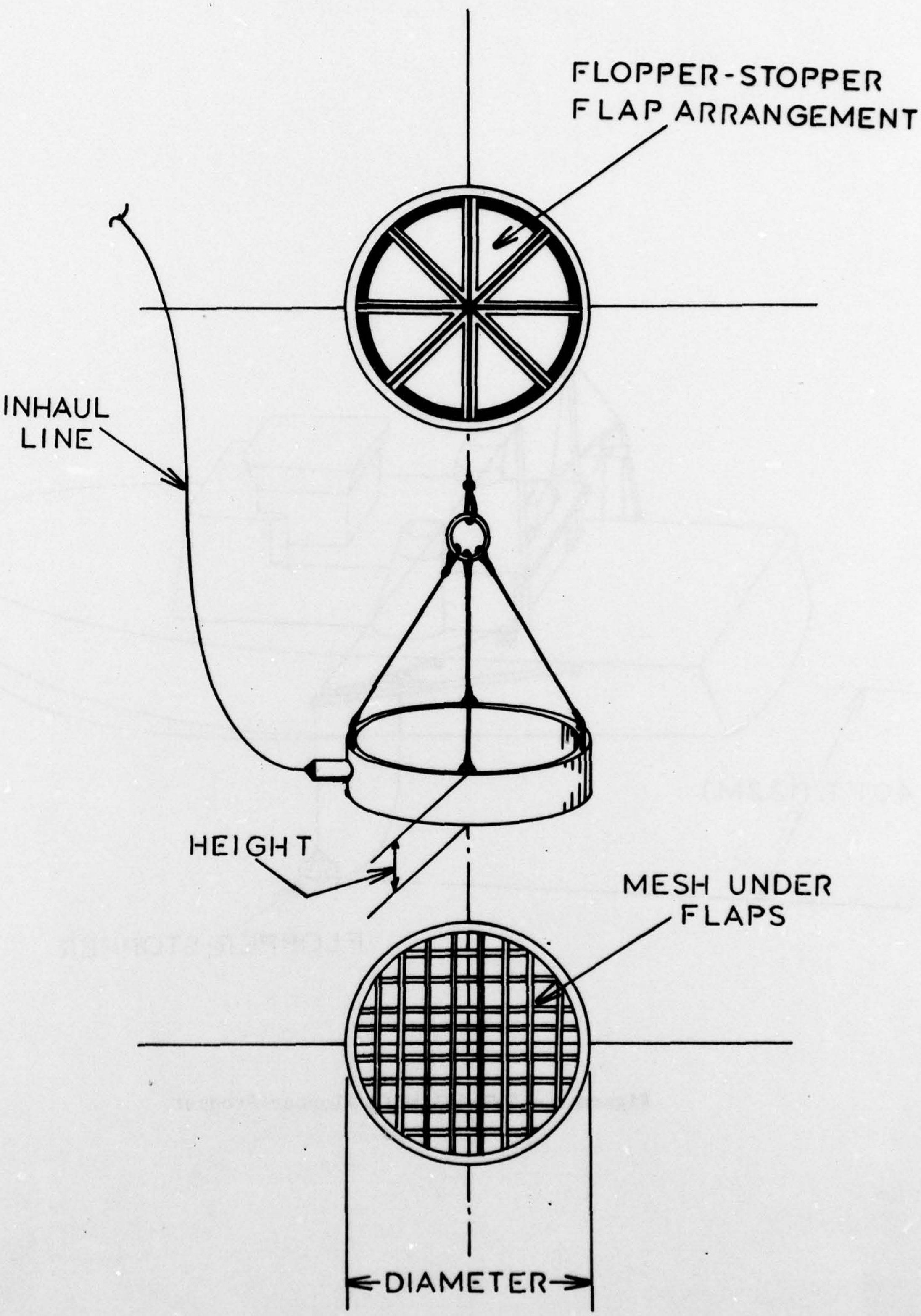


Figure 2 - Details of Flopper-Stopper Bucket

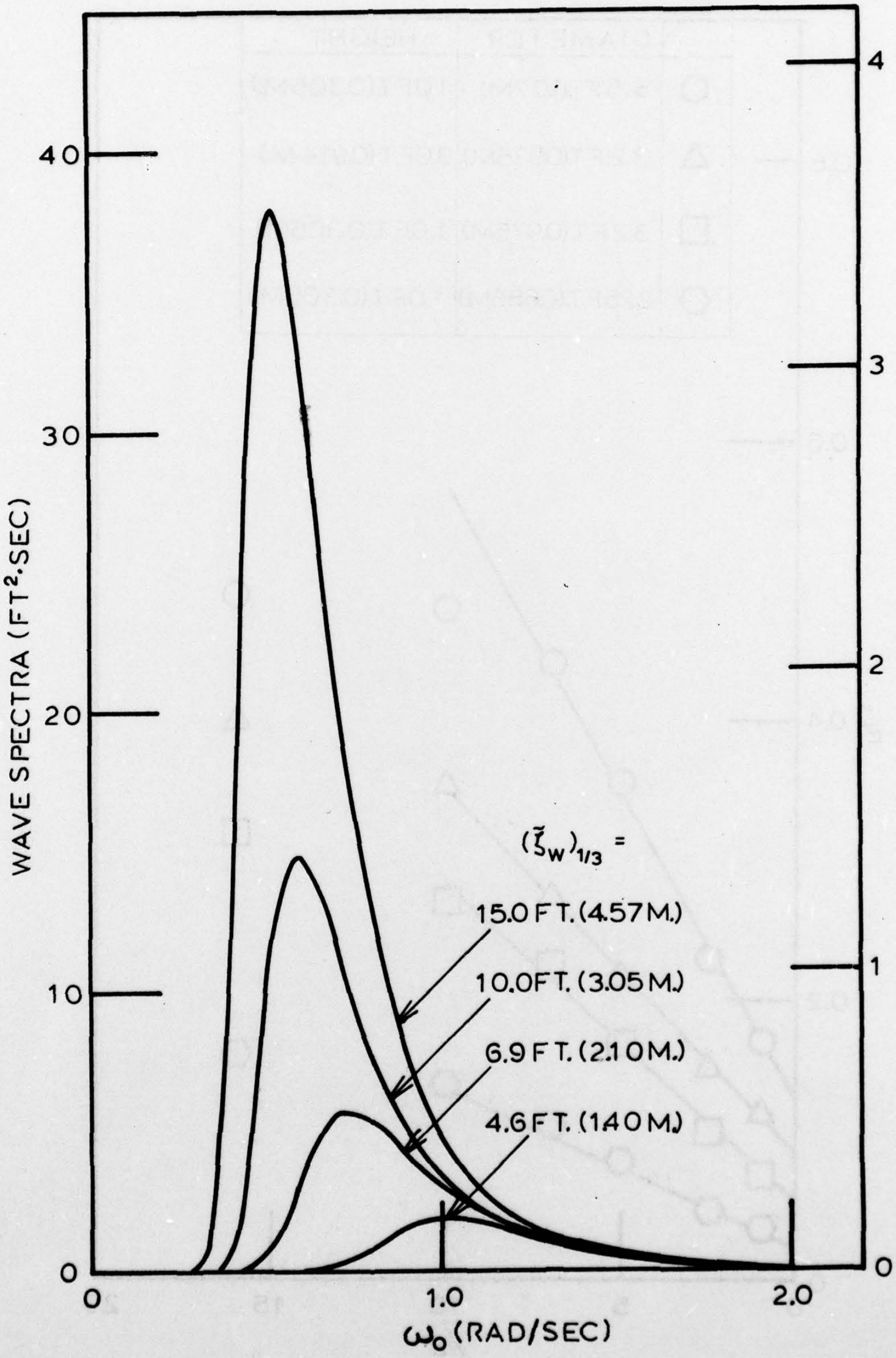


Figure 3 - Sea Spectra

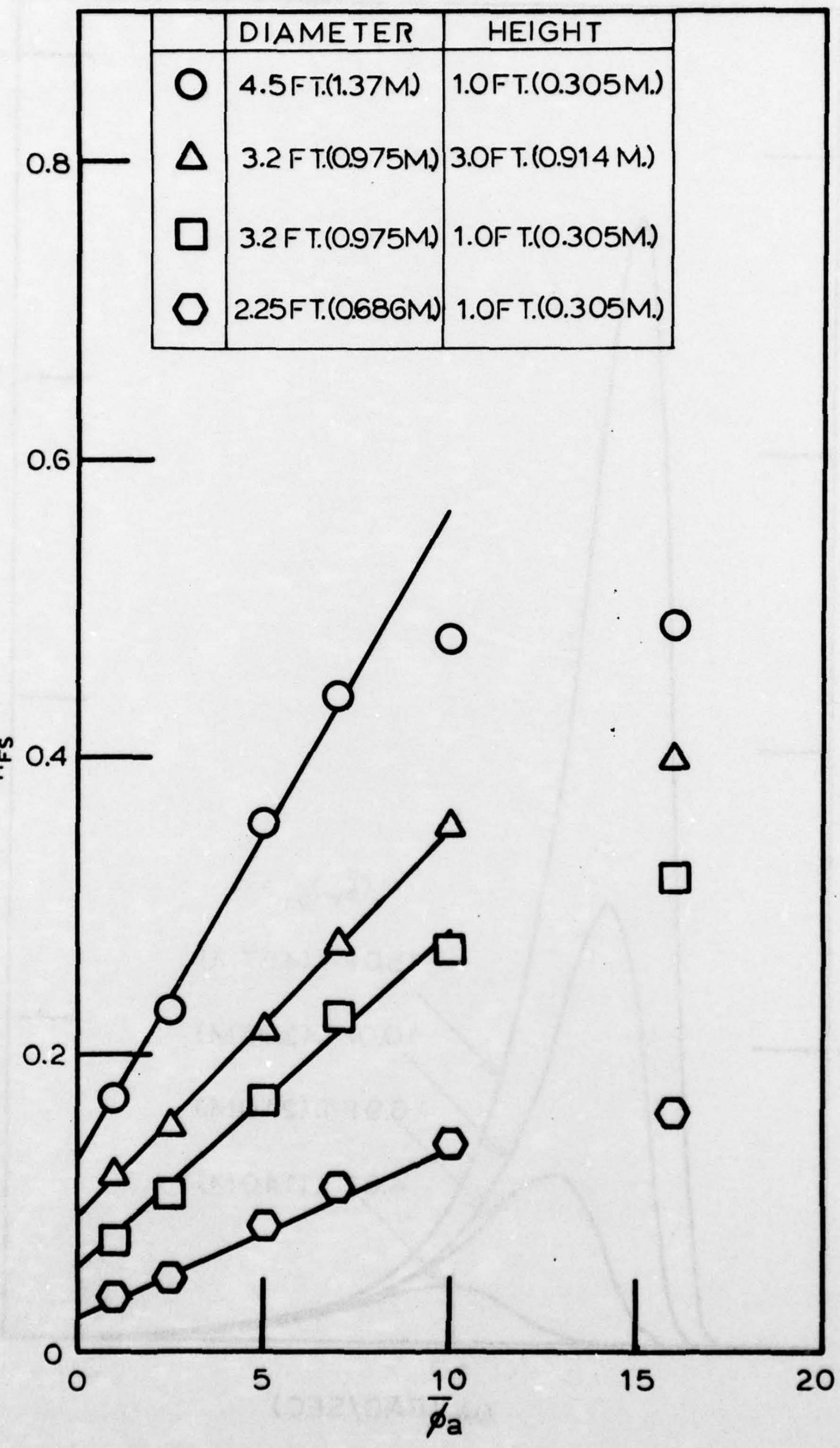


Figure 4 - Flopper-Stopper Roll Decay Coefficients

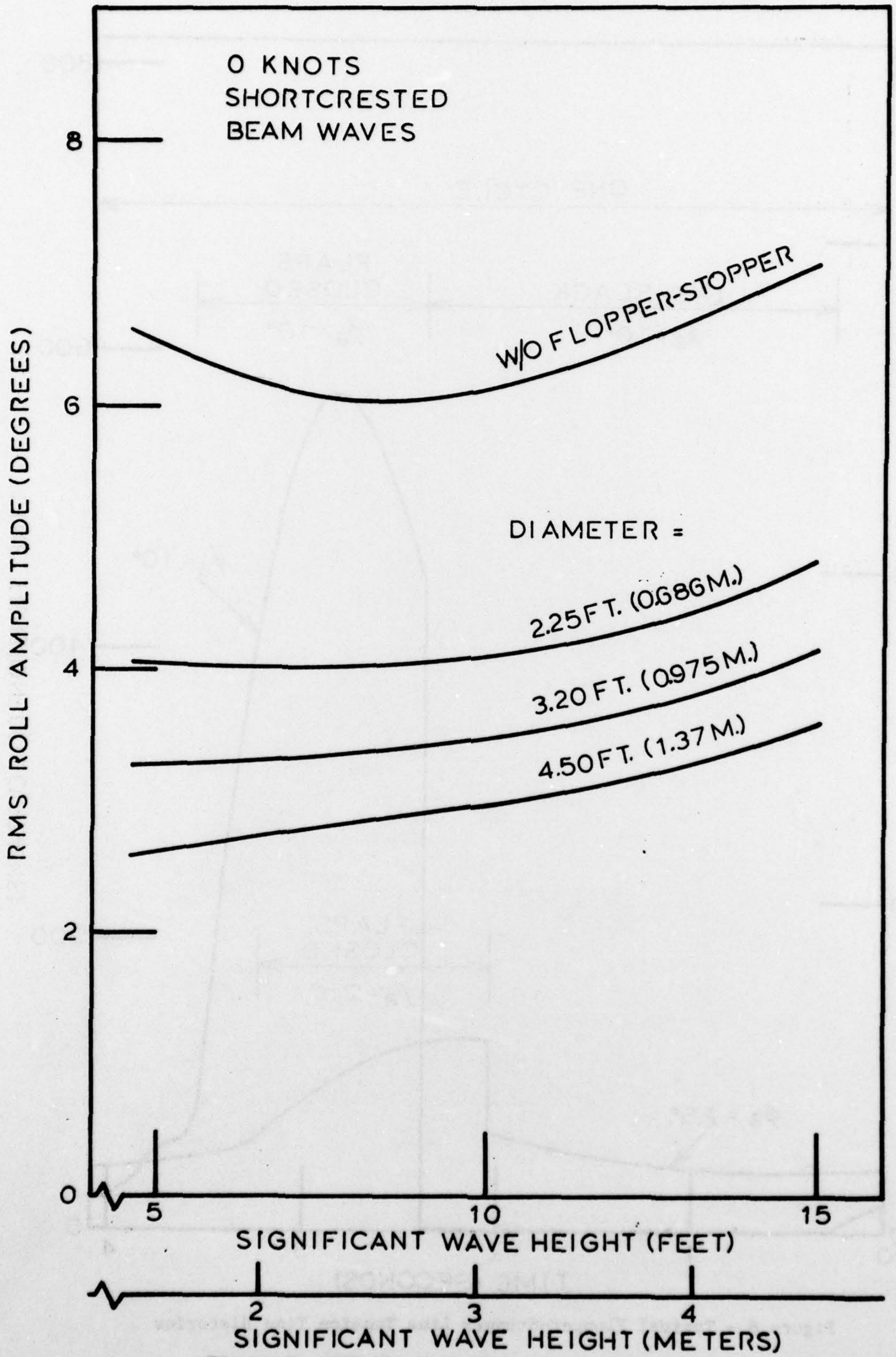


Figure 5 - TWR-681 Root Mean Square Roll Amplitude

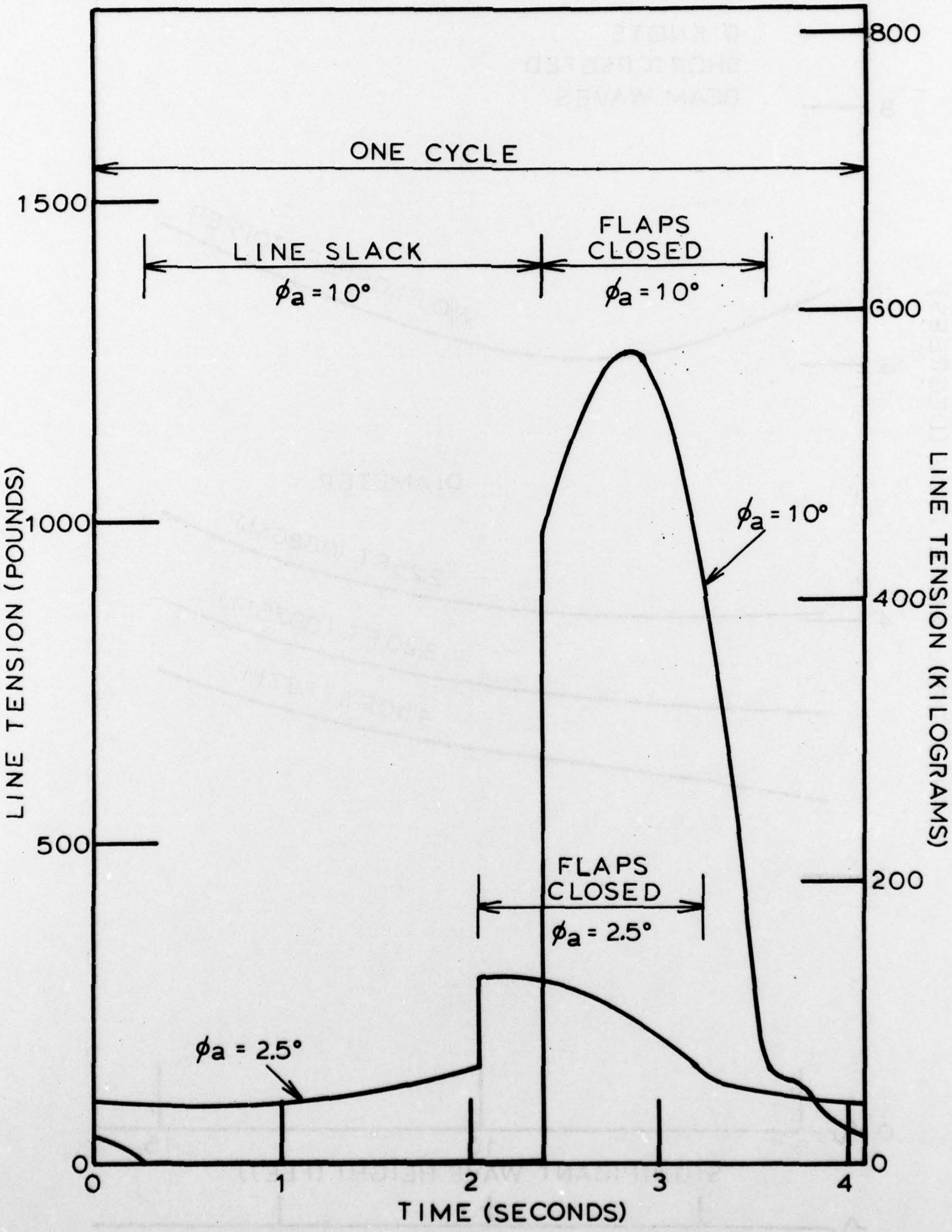


Figure 6 - Typical Flopper-Stopper Line Tension Time Histories

TABLE 1 - SHIP PARTICULARS

Length Over All	100.0 ft (30.5 m)
Length Between Perpendiculars	95.8 ft (29.2 m)
Beam	21.0 ft (6.40 m)
Draft	7.62 ft (2.32 m)
Displacement	339.5 tons (345.0 tonnes)
$\overline{GM}_t$	4.18 ft (1.27 m)
Roll Period	4.05 sec

TABLE 2 - FLOPPER-STOPPER DAMPING

Device Diameter (ft)	Device Diameter (m)	Damping Curve Coefficients a	Damping Curve Coefficients b_e	Limit of Validity (deg)
2.25	0.69	0.022	0.0115	12.5
3.20	0.98	0.054	0.0229	10.0
4.50	1.37	0.123	0.0446	7.5

TABLE 3 - RMS ROLL ANGLE

Sea Condition			RMS Roll Amplitude (Deg)			
Significant Wave Height (ft)	(m)	Modal Period (sec)	Unstabilized	Stabilized Device Diameter 2.25 ft (0.69 m)	3.2 ft (0.98 m)	4.5 ft (1.37 m)
4.60	1.40	6.20	6.57	4.05	3.26	2.59
6.90	2.10	8.50	6.13	4.00	3.33	2.76
10.00	3.05	10.50	6.12	4.10	3.49	2.93
15.00	4.57	11.90	7.06	4.81	4.15	3.62

TABLE 4 - RMS ROLL VELOCITY

Sea Condition			RMS Roll Velocity (deg/sec)			
Significant Wave Height (ft)	(m)	Modal Period (sec)	Unstabilized	Stabilized Device Diameter 2.25 ft (0.69 m)	3.2 ft (0.98 m)	4.5 ft (1.37 m)
4.60	1.40	6.20	10.19	6.28	5.06	4.02
6.90	2.10	8.50	9.51	6.20	5.16	4.28
10.0	3.05	10.50	9.49	6.36	5.41	4.54
15.0	4.57	11.90	10.95	7.46	6.44	5.61

TABLE 5 - RMS ROLL ACCELERATION

Sea Condition			RMS Roll Acceleration (deg/sec^2)			
Significant Wave Height (ft)	(m)	Modal Period (sec)	Unstabilized	Stabilized Device Diameter 2.25 ft (0.69 m)	3.2 ft (0.98 m)	4.5 ft (1.37 m)
4.60	1.40	6.20	15.81	9.74	7.84	6.23
6.90	2.10	8.50	14.75	9.62	8.01	6.64
10.00	3.05	10.50	14.72	9.86	8.40	7.05
15.00	4.57	11.90	16.99	11.57	9.98	8.71

TABLE 6 - RMS DECK EDGE ACCELERATION

Sea Condition			RMS Deck Edge Acceleration							
Significant Wave Height		Modal Period	Unstabilized		Stabilized Device Diameter					
					2.25 ft (0.69 m)		3.2 ft (0.98 m)		4.5 ft (1.37 m)	
(ft)	(m)	(sec)	ft/sec^2	m/sec^2	ft/sec^2	m/sec^2	ft/sec^2	m/sec^2	ft/sec^2	m/sec^2
4.60	1.40	6.20	2.89	0.881	1.78	0.543	1.43	0.437	1.14	0.347
6.90	2.10	8.50	2.70	0.822	1.76	0.537	1.47	0.447	1.47	0.447
10.0	3.05	10.50	2.69	0.821	1.80	0.550	1.54	0.468	1.54	0.468
15.0	4.57	11.90	3.11	0.947	2.12	0.645	1.83	0.557	1.83	0.557

TABLE 7 - CABLE TENSION COEFFICIENTS

Device Diameter		C_T	
(ft)	(m)	lb/deg	N/deg
2.25	0.69	50.7	225.5
3.2	0.98	104.9	466.6
4.5	1.37	218.6	972.3

Note: These values correspond to a 5-degree roll angle.

TABLE 8 - RMS DYNAMIC CABLE TENSION

Sea Condition			RMS Cable Tension					
Significant Wave Height		Modal Period	Device Diameter					
			2.25 ft (0.69 m)		3.2 ft (0.98 m)		4.5 ft (1.37 m)	
(ft)	(m)	(sec)	lb	N	lb	N	lb	N
4.60	1.40	6.20	205.3	913.1	342.0	1520.0	566.2	2518.0
6.90	2.10	8.50	202.8	901.9	349.3	1553.0	603.3	2683.0
10.0	3.05	10.50	210.9	937.9	366.1	1628.0	640.5	2848.0
15.0	4.57	11.90	243.9	1084.0	435.3	1936.0	791.3	3519.0

TABLE 9 - FLOPPER-STOPPER DRAG AND MASS

Device Dimensions				Mass in Air		(Flappers Closed)		(Flappers Open)	
Diameter		Height							
(ft)	(m)	(ft)	(m)	(slugs)	(kg)	$\frac{F_V}{\rho\frac{\pi}{4}d^2V^2}$	$\frac{F_A}{\rho\frac{\pi}{6}d^3A}$	$\frac{F_V}{\rho\frac{\pi}{4}d^2V^2}$	$\frac{F_A}{\rho\frac{\pi}{6}d^3A}$
2.25	0.69	1.0	0.30	2.69	1.22	1.10	1.303	0.36	.022
3.20	0.98	1.0	0.30	4.12	1.87	1.10	1.105	0.34	.012
4.50	1.37	1.0	0.30	6.32	2.87	1.10	0.970	0.32	.007
3.20	0.98	3.0	0.91	10.37	4.70	1.10	2.043	0.42	.012

A = Acceleration, V = Velocity

DTNSRDC ISSUES THREE TYPES OF REPORTS

1. DTNSRDC REPORTS, A FORMAL SERIES, CONTAIN INFORMATION OF PERMANENT TECHNICAL VALUE. THEY CARRY A CONSECUTIVE NUMERICAL IDENTIFICATION REGARDLESS OF THEIR CLASSIFICATION OR THE ORIGINATING DEPARTMENT.

2. DEPARTMENTAL REPORTS, A SEMIFORMAL SERIES, CONTAIN INFORMATION OF A PRELIMINARY, TEMPORARY, OR PROPRIETARY NATURE OR OF LIMITED INTEREST OR SIGNIFICANCE. THEY CARRY A DEPARTMENTAL ALPHANUMERICAL IDENTIFICATION.

3. TECHNICAL MEMORANDA, AN INFORMAL SERIES, CONTAIN TECHNICAL DOCUMENTATION OF LIMITED USE AND INTEREST. THEY ARE PRIMARILY WORKING PAPERS INTENDED FOR INTERNAL USE. THEY CARRY AN IDENTIFYING NUMBER WHICH INDICATES THEIR TYPE AND THE NUMERICAL CODE OF THE ORIGINATING DEPARTMENT. ANY DISTRIBUTION OUTSIDE DTNSRDC MUST BE APPROVED BY THE HEAD OF THE ORIGINATING DEPARTMENT ON A CASE-BY-CASE BASIS.